Operational deflection shape
analysis and vibration solving
for a motion simulator.

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DCT 2006.135
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Chapter 1

Introduction

The Robust Design Engineering Laboratory (RoDEL) at Seoul National University (SNU), South-Korea, has developed a new parallel actuated platform, called Eclipse-II [1]. The platform is used for generating six degree of freedom motions in space. The Eclipse-II platform is very exclusive because it can generate continuous 360 degree rotations around all axes and translations in all directions.

A real size motion simulator, based on the Eclipse-II platform, has been designed and manufactured. During tests, serious vibration issues were found. To be able to give an advice for solving these vibrations, the vibration shapes of the structure under operating conditions are determined. This is done by performing an Operational Deflection Shape (ODS) analysis. When the vibration patterns of the structure are known, it is analyzed which components are critical. Possible design changes are proposed, including adding dynamic vibration absorbers.

This report is organized as follows. Chapter 2 gives an introduction to the Eclipse-II platform. The motion-simulator is presented and the vibration problems are discussed. In chapter 3 the theory on vibration analysis and ODS is explained. Chapter 4 handles the ODS analysis of the motion-simulator. The results of these experiments are presented and discussed in chapter 5. In chapter 6 possible design modifications are proposed. The design of the dynamic vibration absorbers is explained. Moreover, advice for implementing and testing the absorbers on the motion-simulator is given. Finally, some concluding remarks and recommendations follow in chapter 7.
Chapter 2

Eclipse-II

2.1 Eclipse-II Platform

Motion simulators are virtual reality systems. They assume the appearance of a real situation by using movements of a motion-base and audio-visual effects. Most current motion simulators use a hexapod motion base. A big disadvantage of such a base is that it has limited tilt angles (±20 – 30 deg.). Therefore, the Robust Design Engineering Laboratory (RoDEL) has developed a mechanism capable of 360-degree tilting motion as well as translational motion. This parallel platform is called Eclipse-II. For more information about the kinematics of the Eclipse-II platform you are referred to Jongwon Kim et al. ([1] and [2]).

2.2 One-man Ride Machine

After extensive testing of scaled prototypes, a real motion simulator has been designed and manufactured. This machine is called the one-man ride machine[1] and is shown in figure 2.1. Its specifications are given in table 2.1.

Table 2.1: Specifications of the one-man ride machine

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Machine dimensions (L x W x H):</td>
<td>4200 x 4200 x 4700 mm</td>
</tr>
<tr>
<td>Machine weight:</td>
<td>12000 kg</td>
</tr>
<tr>
<td>Kinematic workspace (Ø x H):</td>
<td>236.6 x 491.3 mm</td>
</tr>
<tr>
<td>Max. linear speed:</td>
<td>600 mm/sec</td>
</tr>
<tr>
<td>Max. linear acceleration:</td>
<td>4900 mm/sec² (0.5 g)</td>
</tr>
<tr>
<td>Max. angular speed:</td>
<td>120 deg/sec</td>
</tr>
<tr>
<td>Max. angular acceleration:</td>
<td>500 deg/sec²</td>
</tr>
<tr>
<td>Number of axes:</td>
<td>9</td>
</tr>
</tbody>
</table>

[1] In the hereafter the Eclipse-II one-man ride motion-simulator will be referred to as "one-man ride machine"
2.3 Vibration Issues

During tests, some unexpected vibrations were found. Especially, when the vertical and circular columns are rotating around the vertical axis and high decelerations are applied. In that case the columns are vibrating heavily (see figure 2.2), causing the cabin to vibrate. It should be noted that the vibrations shown in the figure are first assumptions, based on visual observations.

The effects of vibrations in machines can be numerous; noise and annoyance, but also, wear, fatigue, position inaccuracy or even failure (irreversible damage) can occur. Of course, in a virtual reality machine no unwanted vibrations should be present.
3.1 Introduction

Different experimental techniques can be used to analyze vibrations. The most common techniques are experimental modal analysis and operational deflection shape analysis. In experimental modal analysis the exact dynamic behavior of a structure can be determined. This is explained in section 3.2. In operational deflection shape analysis the vibration shapes under real operating conditions can be determined. This is explained in section 3.3.

3.2 Modal Analysis

The dynamic properties of a structure can be determined by FEM modal-simulations (Finite Element Method), or by experimental modal analysis. As the machine already has been built, an experimental approach is chosen. In experimental modal analysis the FRF (Frequency Response Function) of a system is determined. The FRF is a model of a linear system. It is the relationship between the measured output (e.g. displacement) and input (e.g. force), as a function of frequency. When both the applied force and the response to it are measured simultaneously, the FRF can be calculated. From this FRF the natural frequencies, mode shapes and modal damping can be obtained [3].

In modal analysis an important assumption is that the measured structure is isolated from it’s surroundings (e.g. by suspending it by elastic springs), so that no external forces are acting on the system. Considering the size of the one-man ride machine, it is very difficult to isolate it from its environment. Moreover, there is no equipment available to simultaneously apply and measure input forces of this (high) level. Therefore, it is chosen to determine the vibration shapes under operating conditions, using operational deflection shape analysis.
3.3 Operational Deflection Shape (ODS) Analysis

Introduction
In ODS analysis the vibration shapes of a structure, under operating conditions, are determined. The output of the system can be any number of things (displacement, acceleration, etc.). In contrary to modal analysis, the forces acting on the system (inputs) are not measured and, thus, unknown.

The advantages of ODS with respect to modal analysis are [4]:

+ there is no assumption of a linear model
+ the structure experiences actual operating forces
+ true boundary conditions apply

The disadvantages are [5]:

− no complete dynamic model is obtained, so no natural frequencies, mode shapes and damping properties can be determined
− operational deflection shapes only reflect the cyclic motion at a specific frequency, but no conclusions can be drawn for the behavior at different frequencies

In the case of the one-man ride machine, the measured output is acceleration. With these acceleration signals the vibration shapes of the structure can be animated as follows. In specific ODS software the structure is represented by a simple wire-frame. The measurement points are positioned on this frame. The signals are used as inputs for the points on the wire-frame, and shown as (scaled) displacements. For each measured frequency different movements can be animated. This allows the designer to study the vibrations and to identify the critical components. An example of such an animation is given in figure 3.1.

![Figure 3.1: Animation in LMS Test.Lab](image)

**DOFs**
For each measurement point accelerations can be measured in three perpendicular directions. One combination of position and one direction is a Degree Of Freedom, or DOF.
3.3. OPERATIONAL DEFLECTION SHAPE (ODS) ANALYSIS

**Ideal situation**
In the ideal situation, all DOFs for ODS analysis are measured [5]:

1. simultaneously,
2. under constant operating conditions,
3. with signals having a high signal to noise ratio, so that no averaging is required.

Item 1, obviously, depends on the number of DOFs to be measured and on the number of acquisition channels on the measurement system. Item 2 depends on the complexity of the system and whether all DOFs are measured simultaneously or not. When not all DOFs are measured simultaneously, the operating conditions at different measurements can be different. Item 3 depends mostly on the equipment and measurement techniques used.

It is very important that the measurement equipment does not influence the dynamic behavior of the machine. Therefore as little mass, stiffness and damping as possible should be added to the structure.

**Time ODS**
There are three types of ODS [4]:

1. Time ODS
2. Spectral ODS
3. Run-up/down ODS

Time ODS is used to investigate the vibration pattern of a structure as a function of time. Spectral ODS and Run-up/down ODS, in contrary, investigate the vibration pattern at discrete operating frequencies. This is typically done in continuous rotating equipment. Because the vibrations in the one-man ride machine occur at movements with high accelerations and decelerations (e.g. run-stop), time ODS is chosen to be the best method. The measured time signal can be analyzed to see what happens to the machine over time.

**Reference DOF**
In ODS one specific DOF \( j \) is used as reference station. If operating conditions between different measurements change, they hopefully change in the same way for all DOFs, so the change can be canceled out due to the relative measurement. The reference DOF also provides a fixed phase relationship between the different response signals.

**Crosspower spectrum**
When it is assumable that the operating conditions will not change between the different measurement sequences it is possible to just measure the crosspower spectra between each DOF \( i \) and the reference DOF \( j \). The crosspower spectrum is defined as

\[
G_{ij}(\omega) = X_i(\omega)X_j^*(\omega)
\]

where \( X^* \) denotes the complex conjugate [5].

Crosspower functions have the advantage that peaks clearly indicate high response levels. Obviously, crosspower functions are very useful when all DOFs are measured simultaneously.
Chapter 4

ODS on One-man Ride Machine

4.1 Introduction

The measurement procedure and the measurement equipment are described in appendix A. To become familiar with equipment, software and the measurement procedure, a test experiment on a simple plate has been performed. This test experiment is also described in appendix A. The experiments on the machine were repeated once, because the results of the first set of experiments were not satisfying. The major reason was found to be the presence of too much electrical noise. This chapter describes the procedure. Furthermore, the changes of the second experiments with respect to the first experiments are mentioned throughout this chapter to show what was undertaken to improve results.

4.2 Set-up

Setup hardware and define geometry

Considering the high level of electrical noise on the first experiments, the LMS Scadas III 310 measurement system was electrically ground during the second experiments. This reduced noise significantly. Moreover, to enable the simultaneous measurement over 54 channels, an extra data-acquisition system was added.

The LMS Scadas III 310 (described in appendix A.2) was used as master system during both sets of experiments. The LMS Scadas III 316 was added as slave system during the second set of experiments. It is similar to the LMS Scadas III 310, only it has 18 channels in stead of 36. The hardware set-up is shown in figure 4.1. Furthermore, a global coordinate system is defined, see figure 4.2.

![Figure 4.1: Schematic presentation of the hardware set-up](image)
Define DOFs
To define the DOFs to be measured, first, appropriate measurement points are selected. To be able to sufficiently animate the vibrations, measurement points are distributed on the main components. Six points are defined on the circular column and five on each vertical column. Because all movements of the cabin are in direct contact with the user, these should be identified well. In the first set of experiments, these movements were not clearly visible. Therefore, ten measurement points (instead of four) are defined on the platform. Four of them are placed on the frame, containing the cabin. Three points are defined on both the top and bottom surface of the cabin. Furthermore, measurement points were defined on the main joints. In figure B.1 (appendix B) the measurement points on the circular column and on the platform are shown. Also a closer look on some main points is presented. In figure B.2 the measurement points for both vertical columns are shown. Considering the complexity of the machine, it is chosen to measure the acceleration at each measurement point in all three directions (x, y, z).

Define measurement sequence
Considering the number of DOFs to be measured, the experiment is split up in two parts:

1. First, the vibrations of the platform and the circular column are measured (18 points simultaneously, see figure B.1).
2. Second, the vibrations of the two vertical columns are measured (12 points simultaneously, see figure B.2).

The 2 sets of data are combined using the reference DOF (accelerometer number 1).

4.3 Measurements
Setup analyzer
The analyzer software is set-up with the following configuration (There was no reason to change these settings after the first set of experiments). The sample frequency is 2048 Hz,
within a frequency range of 1-1024 Hz. This frequency range is sufficient as low-frequent vibrations are expected. Operating the machine and starting the measurement is done by different persons. A time of 5 seconds ensures overlapping of measurement and operating, so a measurement time of 5 seconds is chosen.

**Calibration and attachment of accelerometers**
After setting up all hardware and making all connections, the accelerometers are calibrated (at the measurement location). Thereby, there is no temperature and humidity difference between the calibration location and the measurement location. The accelerometers are attached to the structure, and local and global coordinate systems are written down.

**Make measurements and validate measurements**
To cause the machine to vibrate, all three motors (two on the vertical columns and one on the circular column) are actuated simultaneously. When the platform is rotated 3.6 degree around \( \theta \) they are stopped. Several trial measurements are done. The crosspower spectrum is calculated for each measurement. When the measurement data seems repeatable and noise seems acceptable, three measurements are done. All data is saved. During postprocessing it is decided which data set is best. It should be noted that, as the technique used is Time ODS, no averaging can be done and no coherence can be calculated to ensure good measurement data.

The accelerometers are relocated to measure the second sequence (i.e. measuring the vibrations of the vertical columns). One accelerometer is kept in place (number one). This reference accelerometer is used to couple the two sets of data.

Results are presented in next chapter.
Chapter 5

ODS Results

5.1 Introduction

The time signal and crosspower spectrum plots of all six measured sequences have been analyzed thoroughly. The best sequences were combined to one set after these analysis. This chapter is based on that data set.

The animation of the geometry, vibrating in time, has been studied to determine the structures' behavior in time during the experiment. The animation at certain frequencies have been given a closer look to determine problematic frequencies. The most pronounced results are presented in this chapter.

5.2 Time Signal

A plot of the time signal is shown in appendix C. Different peaks can be identified in the time signal. A close up on the first three peaks is presented in figure 5.1. In this figure, the first peak (at t = -1.60 s.) is the starting point (when all three motors are started simultaneously). The start-up results in no significant vibrations of the structure. The second and third peaks (at t = -1.24 and -1.21 s.) are caused by the non-simultaneously stopping of the motors. This can be seen when the animation of the time-signal is run.

![Figure 5.1: Time signal of operating area (start-stop)](image-url)
In figure 5.2a the scaled deformation of the structure (in this time animation) at $t = -1.24$ s. is shown. The whole structure seems in normal shape, except the right vertical column, which is deformed heavily. This deformation is caused by the high deceleration applied by the motor attached on that corner.

In figure 5.2b the scaled deformation of the structure at $t = -1.21$ s. is shown. Now, the left vertical column and the circular column are deformed heavily. This is caused by stopping the two concerning motors. As the time signal shows, there is a difference in stopping time of 0.03 second. In the time signal between start and stop peaks, there are relatively low vibration levels. As all motors are started simultaneously, it is expected that stopping all motors simultaneously will decrease the vibration level too. It should be noted that it is *not sure* wether the vibrations will increase or decrease when all motors are stopped simultaneously.

The rest of the time signal seems very chaotic, the structure is vibrating heavily. To analyze if there are repeating movements in the structures, the crosspower spectrum is studied.

![Figure 5.2: Non-simultaneously stopping of the motors](image)
5.3 Crosspower Spectrum

The crosspower spectrum is shown in figure 5.3. Four (low-frequent) peaks can be identified; at 3.65, 5.06, 5.72 and 7.35 Hz. The moving animations of the structure show the vibration shapes at these frequencies. These vibration shapes will be explained here.

1. **3.65 Hertz**
   At this frequency a combination of motions can be identified:
   - Both rectangular columns are vibrating in, what seems to be, their first bending mode.
   - The circular column is vibrating heavily.
   - As a result of these vibrations, the cabin is rotating around $\phi$.

   This peak is clearly the highest peak in the crosspower spectrum.

2. **5.06 Hertz and 5.72 Hertz**
   At these frequencies, especially at 5.06 Hz, the platform is vibrating heavily. It is performing a translational motion in $x$-direction. The vibrations at both frequencies are comparable. It is noticeable that the columns are hardly moving. That is why it is assumed that this movement is a mode of the platform at 5.06 Hz. These peaks are significantly lower than the peak at 3.65 Hz, though.

3. **7.35 Hertz**
   At this frequency the platform translates in $z$-direction. It is noticeable that this frequency is almost exactly twice as high as frequency number 1 (3.65 Hz). Probably it is a harmonic frequency which is excited by the vibrating vertical columns.

All other peaks in the crosspower spectrum have lower amplitudes and are less distinct. Low-frequent vibrations were assumed to form the biggest problem before doing the experiments. These assumptions are confirmed by the results of the experiments. Phase lag between DOFs located at joints has been searched for to detect looseness of joints. No looseness of joints was detected.

![Figure 5.3: The crosspower spectrum](image-url)
Chapter 6

Design Modifications

6.1 Introduction

Chapter 5 made clear that there is a difference in stopping time of the three motors driving the columns. It is unclear whether changing the control algorithms (so that all motors are actuated simultaneously) will result in an increased or decreased vibration level. This should be tested first. It will be done by members of the RoDEL laboratory. When changing the controls does not give the desired effect, this chapter provides other solutions to the vibration problems.

To eliminate or decrease the level of the vibrations, several changes can be considered. The constructions stiffness and inertial parameters can be changed, to tune them against resonances. Damping properties can be increased by using other materials. Finally, dynamic vibration absorbers can be used to reduce the vibration level [6].

6.2 Changing Mass, Stiffness or Damping

The best solution to solve a vibration problem is to separate the components natural frequency from the exciting force frequency. In this case, the high decelerations are required to simulate real-life situations in fighter-aircrafts. Therefore, the exciting forces cannot be changed. The natural frequency can be changed by increasing or decreasing the components mass or stiffness. Furthermore, the vibration level can be decreased by increasing the constructions damping properties, by using different materials. The structural changes to bring the problematic frequencies to another level are expected to be comprehensive. It is expected that the budget of RoDEL laboratory is not sufficient for realizing the required changes. Therefore, it is decided to decrease the vibration problem using Dynamic Vibration Absorbers (DVAs). Nonetheless, some design changes for the next prototype are suggested in appendix D. The design of the DVAs is explained in the next sections.

6.3 Dynamic Vibration Absorber (DVA)

A DVA is a device designed to have the same natural frequency as the (problematic) component it is mounted to. The DVA will vibrate with the same frequency but with a phase difference of 180 degrees, reducing the initial exciting force. The theory behind the working principle of a DVA is explained in Den Hartog [7]. In general, a DVA consists of a simple
mass-spring system; an additional mass, connected by an elastic element to the source of vibration [6].

DVAs have several advantages over changing the construction geometry:
+ DVAs are relatively cheap.
+ While changing geometry can be a matter of trial and error, designing a DVA is a straightforward procedure.
+ Temporary DVAs can be attached. If the temporary absorber does effectively reduce the vibration, it can be left in place till a more permanent solution is designed and installed. If the absorber is not effective, it can be removed with the structure being unchanged.

There is one disadvantage of DVAs:
− If the problem is not resonance, adding a DVA will create a resonance problem. Therefore, if adding a DVA does not give the desired effect, it should be removed again.

Broadly speaking there are two types of DVAs; undamped and damped DVAs. Undamped DVAs work on a narrow frequency band. Damped DVAs work on a wider frequency band, but their effect is less. Moreover, a damped DVA influences the structure at different frequencies [6].

6.4 DVA Requirements

The vertical columns cause the biggest peak in the crosspower spectrum. It is chosen to design a DVA for these columns first. Because the vibrations at 3.65 Hz show a very narrow peak, it is chosen to design an undamped DVA. Before designing the DVA for the vertical columns, clear requirements are set:

• **3.65 Hz**
  The natural frequency should be around 3.65 Hz

• **Tunable**
  The DVA should be tunable, i.e. the natural frequency can be varied in a range (± 10 percent)

• **Safe**
  The DVA should be safe, a lifetime calculation should be made

• **Costs**
  The DVA should be low-cost, so easy to manufacture (drilling, milling), no exotic materials will be used

6.5 DVA Design

To keep the design easy to manufacture a simple design is desired. The natural frequency \( \omega_n \) of the DVA is determined by its mass and stiffness. By simplifying the design, these two parameters are separated. The stiffness is variable by changing the length \( l \) of a cantilever beam, defined as 'spring' (see figure 6.1). The mass, defined as 'block', is not variable. To achieve a tunable natural frequency the block can slide on the spring.

In the following the dimensions in lowercase are dimensions for the spring and in uppercase for the block; \( t \) and \( T \) for thickness, \( w \) and \( W \) for width and \( l \) and \( L \) for length. The thickness
6.5. DVA DESIGN

$T$ of the block is chosen to be 20 times higher than the thickness $t$ of the elastic beam. The resulting stiffness is $20^3$ (see equation 6.4) or order $8 \cdot 10^3$ times higher and can be assumed infinite. Consequently, it can be assumed that the stiffness (read: flexibility) of the system is determined by the spring.

From Korenev et al. [6], it is found that the mass ratio of DVA to structure can lie between 1:5 and 1:20 to give the desired effect. It is chosen to design a DVA with a mass ratio of DVA to structure of 1:10. For the vertical column (including electronics box and part weight of cabin, about 700 kg) this means that the total mass of the DVA should be 70 kg.

Den Hartog [7] gives analytical equations for the simplified model of the DVA. A cantilever beam with a mass $M$ attached to the end has a natural frequency of

$$\omega_n = \sqrt{\frac{k}{M + 0.23m}} \quad (6.1)$$

with stiffness $k$ and mass $m$ of the beam. A natural frequency of 3.65 Hz and a total mass of 70 kg is required. The stiffness of the spring should be

$$k = \omega_n^2 (M + 0.23m) = (3.65 \cdot 2\pi)^2 \cdot 70 = 36.8 \cdot 10^3 N/m \quad (6.2)$$

The stiffness $k$ of a cantilever beam can be calculated by

$$k = \frac{3EI}{l^3} \quad (6.3)$$

with Young’s Modulus $E$ and second moment of inertia $I$ of a rectangular cross section

$$I = \frac{wt^3}{12} \quad (6.4)$$

Looking at equation 6.3 the stiffness is determined by the parameters $E$, $I$ and $l$.

Material choice

To achieve a low natural frequency, low stiffness is required. The allowable strain for a material is defined by $\sigma_y/E$. It gives an indication of how much deformation the material can tolerate. The properties of carbon steel, spring steel and aluminium are given in table 6.1. Aluminium has the lowest Young’s Modulus. Moreover, it has the highest $\sigma_y/E$-value. Spring steel also has good properties, but it is not easy to manufacture. Therefore, aluminium is chosen. An extra advantage of aluminium is that it hardly corrodes.
Table 6.1: Material properties

<table>
<thead>
<tr>
<th>Material</th>
<th>(\rho) ([kg/m^3])</th>
<th>(E) ([GPa])</th>
<th>(\sigma_y) ([MPa])</th>
<th>(\sigma_y/E)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon steel</td>
<td>7830</td>
<td>207</td>
<td>500</td>
<td>(2.4 \cdot 10^{-3})</td>
</tr>
<tr>
<td>Spring steel</td>
<td>7830</td>
<td>207</td>
<td>1000</td>
<td>(4.8 \cdot 10^{-3})</td>
</tr>
<tr>
<td>Aluminium</td>
<td>2800</td>
<td>70</td>
<td>500</td>
<td>(7.1 \cdot 10^{-3})</td>
</tr>
</tbody>
</table>

**Dimensions block**

To realize a mass of 70 kg a volume of \(m = \rho \frac{m}{\rho} = 0.25 m^3\) of aluminium is required. To achieve low stiffness of the cantilever, the length \(l\) should be maximized. Smart placement of the DVA is necessary. The total length of the absorber \(L_{tot} = l + L\) can not exceed 750 mm, as that is the maximum space between the vertical column and the frame. This place is chosen for mounting the DVA (which will be shown in section 6.6). To leave enough space for the length of the spring, the length \(L\) of the block is set to 200 mm. As mentioned, the total thickness \(T\) is 20 times the thickness of the spring, or 300 mm. The resulting width \(W\) to achieve a volume of 0.25 \(m^3\) is 417 mm.

**Dimensions spring**

Setting the initial length \(l\) of the spring to 500 mm leaves 20 mm outward movement of the block for decreasing the natural frequency and 30 mm space between DVA and frame. As will be shown later, the fatigue stress depends on the surface properties of the material. A smooth surfaces will fail at a higher number of cycles than a rough surface with notches and cracks. Therefore, it is chosen to use aluminium of a standard thickness. This standard rolled plate has good surface properties which should not be altered by using cutting techniques. The sides of the plate should be finished using peripheral milling. It is very important that the sides are machined using down milling (or climb milling), so the chips will not damage and roughen the finished surface.

It is found that aluminium plate is available in standard thicknesses of 10 and 15 mm. Now, the dimensions of the spring for a thickness of 10 and 15 mm are determined. The following parameters are known; \(k\), \(E\) and \(l\). Equation 6.3 written explicit for \(I\) gives

\[
I = \frac{k l^3}{3E} = \frac{36.8 \cdot 10^3 \cdot 0.5^3}{3 \cdot 70 \cdot 10^9} = 2.19 \cdot 10^{-8}
\]

The maximum stress in the spring can be calculated when the maximum force at the end of the spring is known. From tabel 2.1 we know that the maximum angular acceleration of the platform is 500 \(deg/sec^2\) or \(2\pi/360 \cdot 500 = 8.73 rad/sec^2\). The distance (radius) between mass and axis of rotation (z-axis) is approximately 2.4 m. This gives a maximum linear (tangential) acceleration of \(2.4 \cdot 8.73 = 20.95 m/sec^2\). The bending force \(F\) on the spring is then \(F = m \cdot a = 70 \cdot 20.95 = 1467N\).

From Fenner [8], we find that the maximum stress in a cantilever beam of length \(l\) with a concentrated force \(F\) at the end is

\[
\sigma_{max} = \frac{FLt}{2I}
\]

\[\text{(6.6)}\]
6.5. DVA DESIGN

For a thickness of 10 mm we find

\[
\sigma_{\text{max}} = \frac{1467 \cdot 0.6 \cdot 0.010}{2 \cdot 2.19 \cdot 10^{-8}} = 201 \text{MPa} \tag{6.7}
\]

For a thickness of 15 mm we find

\[
\sigma_{\text{max}} = \frac{1467 \cdot 0.6 \cdot 0.015}{2 \cdot 2.19 \cdot 10^{-8}} = 301 \text{MPa} \tag{6.8}
\]

Fatigue life of spring

The fatigue stress is a value for the allowed stress in a material to withstand a number of cycles until failure. It depends on material properties, the surface properties and on the number of load cycles. To be able to guarantee safety over a sufficient test period, the number of load cycles is determined:

It is expected that the DVA vibrates over a maximum time of five seconds per stop. This means a maximum of 20 vibrations (at 3.65 Hz) or load cycles per stop. The DVA will be designed for 1000 start/stop-runs. This is sufficient to test the DVA extensively. 1,000 runs equals 20,000 cycles. A safety factor of five is chosen to guarantee safety over the test period. Failure should not occur before completing 100,000 cycles. It should be noted that the spring should be removed after 1000 start/stop-runs, though.

The most common way to express fatigue stress is with SN-curves. In SN-curves the number of cycles at which failure occurs is plotted against the maximum occurring stress in these load cycles. In figure 6.2 a SN-curve for aluminium is shown. As can be seen in the figure, aluminium parts with a smooth surface have a longer lifetime than parts with a rough surface. Plate of standard thickness and peripheral milled sides is used. The figure shows that the maximum allowed stress for a lifetime of $10^5$ cycles, with a normal to smooth surface is 30 ksi, or 207 N/m². Comparing this value with the maximum stress calculated in 6.7 and 6.8 shows that only the spring with a thickness of 10 mm satisfies the lifetime calculation.

![Figure 6.2: SN-curve for aluminium](image)
CHAPTER 6. DESIGN MODIFICATIONS

Realization
Using equation 6.4 we can find that for a thickness \( t \) of 10 mm the width \( w \) of the spring becomes 263 mm. The width \( w \) of the spring is much higher than the thickness \( t \). This ensures that the mass will vibrate in the desired (flexible) direction. The total length \( L_{tot} \) extending from the column is 720 mm. It is composed as follows; working length spring \( l \) 500 mm, extension for clamping the block 200 mm, extension for tunability 20 mm. The plate is extended for 200 mm inwards so it can be mounted to the column. The total length becomes 920 mm. Four 12 mm holes are drilled in the plate, which can contain 12 mm threads to mount it to the column.

The block is divided into two parts. Each block contains four 12 mm holes, and is clamped on the spring with 12 mm threads. The blocks can slide on the spring to change the natural frequency of the DVA. The only manufacturing techniques that should be used are drilling and milling.

In table 6.2 the dimensions for spring and block are shown. A CAD drawing of the DVA is shown in figure 6.3. Technical drawings can be found in appendix E. It is advised that the blocks are connected to the column with a steel cable, for extra safety. This should be included in the design.

<table>
<thead>
<tr>
<th>( t/T ) [mm]</th>
<th>( w/W ) [mm]</th>
<th>( l/L ) [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring</td>
<td>10</td>
<td>263</td>
</tr>
<tr>
<td>Block</td>
<td>200</td>
<td>417</td>
</tr>
</tbody>
</table>

Figure 6.3: CAD drawing of the assembled DVA
6.6 DVA Implementation and Tuning

For safety, it is very important that the DVA\(s\) are fixed well. It is advised that the blocks are connected to the column with a steel cable. Damping decreases the effect of the DVA \cite{6}. Therefore, a hysteresis-free connection should be achieved. This can be achieved by bolting the DVA to the vertical column. Four 12 mm holes should be drilled in each vertical column. In figure 6.4 an example for the implementation of the DVA for the vertical column is shown.

![DVA mounted to vertical column](image)

Figure 6.4: DVA mounted to vertical column

When the DVAs are attached, a first run/stop can be operated. Visual inspection should be done to check if the initial vibration problem has decreased or increased.

**Iterative test procedure**

Because the intensity of the exciting force is not known, some trial-and-error experiments might be needed. The DVA should be tuned, such that its natural frequency exactly matches the natural frequency of the vertical column. This can be done by repeating this sequence:

1. run/stop the machine
2. check the effect of the DVA
3. move the DVA on the spring
4. start at number 1.

If improvements are visible, ODS can be performed again to verify the observations. If no improvements are seen, the DVA should be removed, as it can introduce a resonance in the structure.
Chapter 7
Conclusions and Recommendations

Two options have been considered for analyzing the vibrations in the one-man ride machine. Experimental modal analysis was rejected because the proper equipment was not available. Moreover, the size of the machine makes it impractical to isolate it from its surroundings. Operational deflections shape analysis was expected to be the best method for analyzing the vibration shapes of the motion simulator.

Two sets of ODS experiments were performed, resulting in clear and satisfying results. During the experiments, all three columns were rotated around the vertical axis, and stopped. The time signal shows that one of three motors (attached to the right vertical column) is stopped 0.03 seconds too early. The results show that the machine suffers from serious, low-frequent, vibrations. The crosspower spectrum of the measurements is analyzed to identify the problematic components. The highest peak indicates that the vertical columns are vibrating heaviest at 3.65 Hz. This results in vibrations of the cabin at twice that frequency, 7.35 Hz. The cabin is vibrating separately around 5.06 Hz.

Several options for modifying the one-man ride machine have been considered. Structural changes (moving inertia or adding stiffness and/or damping) are expected to be effective but too comprehensive and, thus, too expensive. Nevertheless, several design changes are proposed which should be considered when building a second prototype. Tunable dynamic vibration absorbers for the vertical columns are designed. They are designed to decrease the vibrations of these columns at 3.65 Hz. An advice is given to implement the DVAs. They are designed for a lifetime of 1000 start/stop-runs, which should be sufficient to test the machine extensively. Technical drawings are included in the appendices.
Recommendations

- The control algorithms should be changed, such that all three motors are actuated simultaneously.

- If changing the control algorithms does not decrease the vibrations (at high decelerations), dynamic vibration absorbers can be attached to the vertical columns.

- It is very important that the production methods described in this report are used. The sides of the spring should be finished using peripheral (or climb) milling.

- An extra connection between blocks and column is advised. A steel cable can be used. This is not included in the design.

- If the DVAs are effective, new ODS experiments should be performed to analyze the behavior of the construction.

- Before building a second prototype, a thorough dynamic (modal) analysis should be performed. In appendix D, several design changes are proposed to improve the dynamic behavior of the machine.

- This report might be used as a reference for performing ODS experiments.

- This report might be used as a reference for designing DVAs.
Bibliography


Appendix A

Equipment, Procedure and Test Experiment

A.1 Introduction

In this appendix the equipment used for ODS experiments is presented. The equipment has been made available by the Acoustics & Vibrations Laboratory of the Advanced Automotive Research Center at SNU. The measurement procedure is divided in different steps, providing a clear plan to do the experiments. To become familiar with equipment, software and the measurement procedure, a test experiment on a simple plate has been performed.

A.2 ODS Equipment

Data acquisition front-end
The *LMS Scadas III 310* signal conditioning and data acquisition system is a 36-channel portable lab-device. It is shown in figure A.1. The system enables simultaneous measurement of 36 DOFs. It is connected to a PC through a PCMCIA-card.

Figure A.1: The *LMS Scadas III 310* data acquisition system
APPENDIX A. EQUIPMENT, PROCEDURE AND TEST EXPERIMENT

Software
The software for data acquisition, analysis and visualization of results is LMS Test.Lab. This software forms a perfect combination with the LMS Scadas III data acquisition system. The whole procedure of calibration, defining the measurement range, doing measurements, computing results and displaying animations is integrated in one software package.

Accelerometers
The accelerometers are Brüel & Kjaer type 4506. These are piezoelectric triaxial accelerometers with a low mass compared to the structure of the machine. The specifications are given in table A.1. Each accelerometer is calibrated in all three directions before they are attached. They are mounted on the structure using Endevco accelerometer mounting wax or Henkel 401 bond.

Table A.1: Specifications of B&K 5406 accelerometer

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensitivity</td>
<td>about 10.00 mV/ms⁻² (is calibrated before each measurement)</td>
</tr>
<tr>
<td>Frequency range</td>
<td>1 - 6000 Hz</td>
</tr>
<tr>
<td>Weight</td>
<td>15 g</td>
</tr>
</tbody>
</table>

Vibration calibrator
The device used for calibrating the accelerometers is a Brüel & Kjaer 4294 exciter. It has a reference vibration level of 10 ms⁻². The operating frequency is fixed at 159.15 Hertz (= 1000 rads⁻¹). It is shown in figure A.2. The measured vibration level is used as a reference for the accelerometer signal.

Figure A.2: The B&K 4294 exciter with the B&K 4506 accelerometer mounted to it
A.3 Measurement Procedure

This section provides a clear measurement procedure which was followed narrowly to prevent mistakes. The stepwise measurement procedure is divided in three main parts and is advised by Brüel & Kjær [4]:

1. Setup:
   - **Setup hardware**
     First, all hardware should be set up. All connections should be made and tested.
   - **Define geometry**
     The geometry of the test object should be well-defined, so that the DOFs to be measured can be chosen.
   - **Define DOFs**
     Before attaching the accelerometers to the structure, suitable DOFs should be determined. A rule of thumb is that there should be something (acceleration) to measure on that specific position in that specific direction. To detect looseness of joints, either sides of joints are also good measurement points.
   - **Define measurement sequence**
     In the case that there are less input channels (on the data acquisition system) then DOFs to be measured, a measurement sequence should be chosen. It is very important that operating conditions do not change between the different measurements. As Time ODS is used, averaging is not possible.

2. Measurements:
   - **Setup analyzer**
     First step of setting up the analyzer consists of choosing the frequency range and resolution. Time signals and crosspower spectra should be measured. All DOFs should be linked to a different channel in the ODS software.
   - **Calibration**
     The transducers are calibrated using the exciter described in section [A.2]. Therefore the accelerometer is placed on the exciter. The ODS software has an auto-calibration function, which uses the vibration level of the exciter as a reference.
   - **Attach accelerometers and make trial measurement**
     The accelerometers are attached using Henkel bond. This provides a reliable, light and stiff connection. The wires should not be entirely fixed to the structure. When the wires hang loose, they vibrate less, thereby generating less noise in the signal.
   - **Make measurements**
     When all transducers are calibrated and well-fixed and the measurement range is chosen, the measurement can start. The machine is operated at normal, but problematic, accelerations.
   - **Validate measurement**
     After measuring, the data should be verified. When the results seem not satisfying, both the equipment and the set-up should be checked and the measurement should be repeated.
3. Results and documentation:

- Transfer measurements to animation program
  When the measurement is successful the data can be transferred to an animation program. In this case that is another module in the same software package. The DOFs can be linked to the corresponding directions in the nodes of the defined geometry. For different frequencies the response of the structure can be visualized.

- Geometry animation
  Now, the designer can analyze the vibration animation and can decide which components are critical in what direction. Depending on this information, modifications in the design of the critical components can be done to reduce vibrations.

A.4 Test Experiment: ”Modal Analysis of a Simple Plate”

Introduction
Before going to Incheon (location of the one-man ride machine) to do the ODS experiments, a test experiment is performed. In this way one becomes familiar with the software and equipment before the real experiment is performed. Moreover, the experimental procedure can be practiced.

Modal analysis of a simple plate
To keep the practice simple and verifiable it is chosen to do the test experiment on a simple plate. Because a simple plate does not have operating conditions it is chosen to do a modal analysis experiment. However, the same equipment and software is used. The plate is isolated from its environment by attaching it to a frame through weak springs (see figure [A.3]).

Figure A.3: The test experiment set-up
When the analyzer is set, the geometry is defined. A grid of six points on the plate’s surface is defined. Measurement points are chosen on locations away from nodes of modes that are expected. Each of the six accelerometers is calibrated in all three directions (x, y, z) and positioned on the specified points, with the corresponding orientation (see figure A.4).

Now, the trigger for hammer-impact can be set. The impact frequency range is chosen by impacting several times. When all ranges are well-defined, the roving hammer experiment can begin. To reduce noise, for each point an average of five impacts is taken. When all six points are excited, the results are analyzed. The accelerations can be transformed to displacement signals by integrating twice. These displacements can be linked with points on the wireframe, thereby animating the vibrations of the plate. The first six natural frequencies can easily be recognized in the FRF (see figure A.5). The first six modes (three bending and three rotation modes) can be visualized in the animation. After performing the test experiment, the equipment is known well and the measurement procedure is clear.

Figure A.5: In the FRF of the test experiment the first six natural frequencies can easily be recognized
Appendix B

Measurement Points

Figure B.1: Measurement points for first sequence
Figure B.2: Measurement points for second sequence
Appendix C

Time Signal

Figure C.1: Plot of the time signal that was selected best
Appendix D

Design changes next prototype

If, in the future, a second prototype of the one-man ride machine is built, these recommendations for changes in the design should be taken in mind:

- All unnecessary inertia should be removed, the keywords are ‘light’ and ‘stiff’.
- The horizontal sections of the vertical columns are over-dimensional in z-direction. The weight of the columns can be supported by the wheel guiding at the bottom of the structure.
- Currently, the cabin is connected to the platform frame with low rotation stiffness around $\psi$. This connection can be made stiffer by moving the link with the circular column to the top of the frame containing the cabin.
- The weight of cabin + platform frame is about 800 kg. This should be reduced (e.g., by using sandwich constructions, they are light-weight, very stiff and have good noise properties).
- The electronics boxes should be mounted as close to the motors as possible. This means that the box is mounted low on the vertical column. Moreover, the electronics boxes become better accessible for people.
- The circular column’s stiffness should be increased in transverse direction and decreased in radial direction. There are low forces in the radial direction, which is dimensioned stiffest. The main driving forces on that column are tangential, and the inertial forces are in transverse direction.
APPENDIX D. DESIGN CHANGES NEXT PROTOTYPE
Appendix E

Technical drawings

On the next two pages, the technical drawings for blocks and spring can be found. These drawings are downscaled from A3 size. The block is on a scale of 1:2 and the spring on a scale of 1:5.
Figure E.1: Technical drawings block
Figure E.2: Technical drawings spring